



# RESEARCH MEMORANDUM

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ROTOR ON THE PERFORMANCE OF A CONSERVATIVELY  
DESIGNED TURBINE

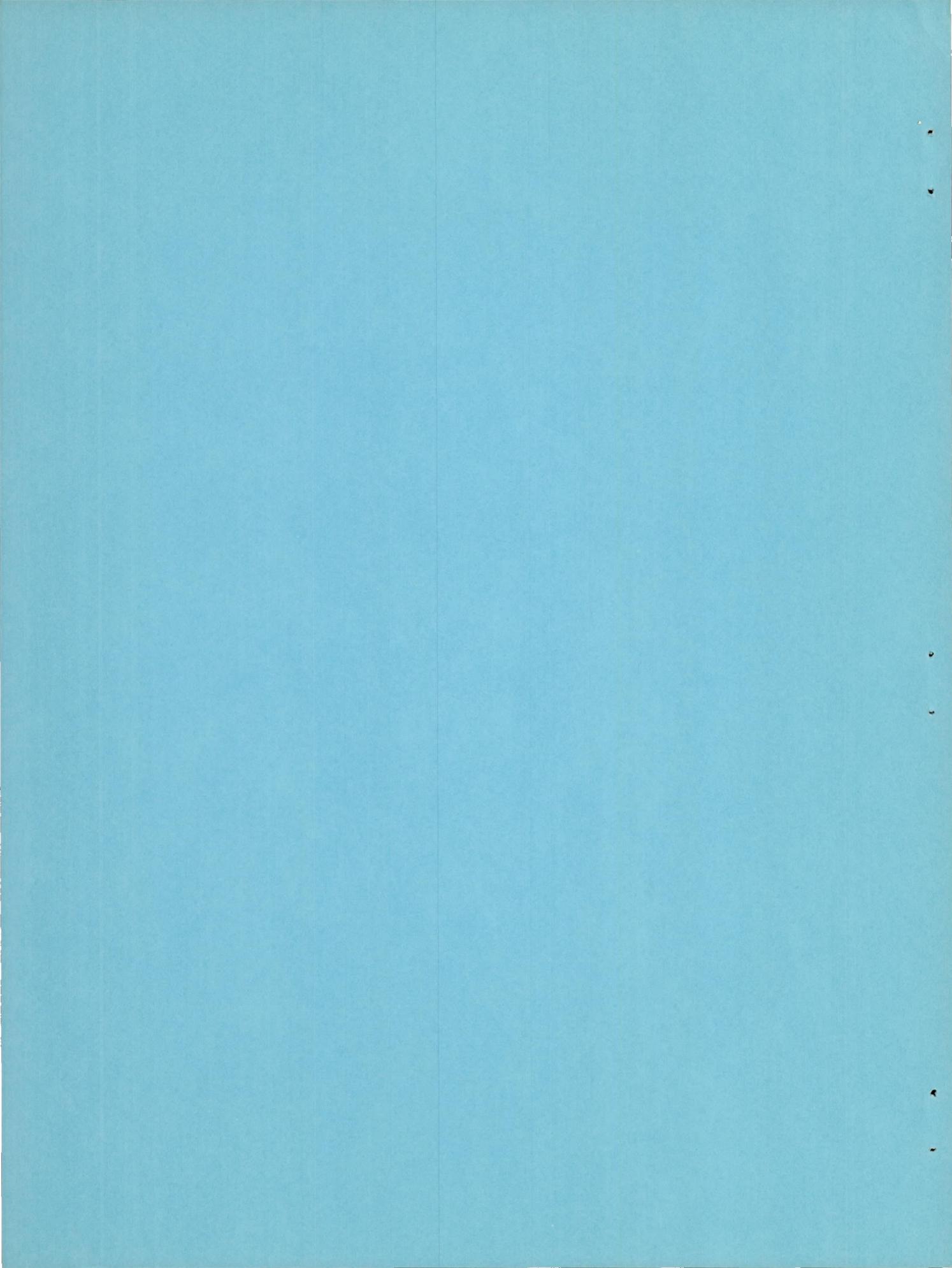
By Cavour H. Hauser and Henry W. Plohr

Lewis Flight Propulsion Laboratory  
Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE  
FOR AERONAUTICS

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EXPERIMENTAL INVESTIGATION OF THE EFFECT OF A SHROUDED ROTOR  
ON THE PERFORMANCE OF A CONSERVATIVELY DESIGNED TURBINE

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SUMMARY

In order to determine the effect of shrouding the rotor blades of a particular turbine design on the over-all turbine performance, a conservatively designed experimental cold-air turbine was investigated both with and without a shroud band on the rotor blades. The tip of the rotor blade was oriented toward the tangential direction so that the adverse effect of blade-tip scraping would not be appreciable. It was found that the addition of a shroud to the turbine-rotor blades had only a slight effect and did not improve the turbine performance. The efficiency for the unshrouded-rotor configuration was about one point higher than for the shrouded rotor.

INTRODUCTION

The use of shrouding on the rotor blades of aircraft gas turbines affects both mechanical and aerodynamic aspects of the design. If light, high-aspect-ratio rotor blades are used, the use of a shroud becomes necessary in order to dampen blade vibrations. This advantage must be balanced against the problem of maintaining the blade-root stress at a reasonable value while still supporting the additional weight of a shroud. Aerodynamically, it has previously been considered that the primary effect of shrouding was to reduce the leakage across the blade tips from the entrance to the exit of the blade row. With the shroud submerged in the outer casing, the resistance offered to this leakage flow would be increased. However, recent studies of the secondary flows in the tip region of rotor-blade rows have shown that other effects are probably of greater importance (ref. 1). Three separate secondary-flow effects were found to occur, as illustrated in figure 1: (1) There is cross-passage flow in the boundary layer along the outer annular wall of the turbine from the pressure to the suction surface of the blades. This flow forms a vortex referred to as the "passage vortex". (2) When tip clearance exists, the leakage flow passing over the blade tip from the pressure to the suction surface forms a "tip-clearance vortex". (3) Finally, the motion of the rotating blade row has a "scraping effect"

tending to scrape boundary-layer air from the stationary outer-annulus wall and mix this low-energy air with the main flow passing through the blade row. For unshrouded rotor blades, the pressure gradients giving rise to the blade-passage vortex and the blade scraping effect may be balanced, to some extent at least, by the tip-clearance vortex effect. Also, if the blade-tip profile is oriented toward the tangential direction with a negative relative blade-entrance angle  $\beta$  (fig. 1), the boundary layer will tend to be sliced rather than scraped from the outer-annulus wall, thereby decreasing this particular component of loss. It is therefore possible that under certain conditions the total loss associated with an unshrouded-rotor-tip configuration will be less than that for the case of the shrouded rotor.

In order to evaluate the effect of shrouding the rotor blades of a particular turbine design on the over-all turbine performance, an experimental investigation has been made at the NACA Lewis laboratory. The turbine-rotor blading used in this study is that illustrated in figure 1, having the tip oriented in the tangential direction and thus having a favorable scraping action on the wall boundary layer. The performance of this conservatively designed turbine was first obtained for the unshrouded-rotor configuration (ref. 2). A shroud was then applied to the rotor blades and the performance obtained for this configuration. The comparative results of these performance studies are presented.

#### DESCRIPTION OF TURBINE AND EXPERIMENTAL PROCEDURE

The aerodynamic design of the turbine used in this investigation is given in detail in reference 2. The following specifications are repeated herein (all symbols are defined in the appendix):

Equivalent weight flow, $w\sqrt{\theta_1}/\delta_1$ , lb/sec . . . . .	16.60
Equivalent design work, $\Delta h/\theta_1$ , Btu/lb . . . . .	16.14
Over-all stagnation pressure ratio, $p'_1/p'_3$ . . . . .	1.751
Equivalent mean blade speed, $U_m/\sqrt{\theta_1}$ , ft/sec . . . . .	625
Turbine hub-tip ratio, $r_h/r_T$ . . . . .	0.60
Turbine outer diameter, $d_T$ , ft . . . . .	1.167

The turbine was designed with free-vortex flow and simple radial equilibrium assumed at the exit of both the stator and rotor blades. The rotor-blade-profile design is shown in figure 2.

The unshrouded turbine rotor considered herein is the 44-blade-rotor design discussed in reference 2. For the shrouded-rotor configuration, a continuous steel band was thermally shrunk on the perimeter of the existing 44-blade rotor. Figure 3 is a photograph of the shrouded-rotor configuration being installed in the test facility.

Because the shroud ring was shrunk over the wheel with considerable mechanical interference to prevent it from slipping out of place during operation, the rotor blades were distorted a slight amount from the force of the shrunken shroud ring. However, the average measured rotor-blade-throat opening varied less than 1 percent before and after the shroud ring was assembled with the rotor, and therefore it was considered that the effect of the distortion was negligible.

A diagrammatic comparison of the shrouded- and unshrouded-rotor configurations is shown in figure 4. The tip clearance was held constant at 0.030 inch for both the shrouded and unshrouded configurations. The thickness of the shroud band was 0.090 inch. As shown in the diagram, the outer casing was faired between the stator and rotor for the shrouded-rotor configuration so that the air could flow smoothly from the stator into the rotor at the tip section.

The same experimental procedure was followed in determining the performance of each of these turbine configurations and is given in detail in reference 2. Over-all performance data were taken at nominal values of stagnation pressure ratio  $p_1'/p_3'$  from 1.20 to the maximum obtainable while the wheel speed was varied from 0.60 to 1.10 of equivalent design speed in 0.05 intervals (7800 to 14,400 rpm for an inlet temperature of 540° R).

The brake internal efficiency, which is based on expansion between the entrance and exit stagnation pressures, was used to express turbine performance. This efficiency is defined as  $\eta_t = E/(h_1' - h_3')$  where  $E$  is the measured turbine shaft work. The ideal enthalpy drop  $(h_1' - h_3')$  was computed from the values of entrance and exit stagnation pressure and entrance stagnation temperature. The exit stagnation pressure was computed by adding to the measured static pressure a dynamic pressure corresponding to the axial component of the exit velocity computed from continuity considerations. An average measured exit temperature was used in these calculations.

Detailed exit flow surveys of the shrouded-rotor configuration were made in order to compare the survey results of the unshrouded- and shrouded-rotor configurations. It was desired to measure any shift in the measured distribution of losses at the rotor exit. Surveys behind rotating blades are inherently difficult because of the rapid fluctuations in the flow caused by passing blade wakes. In regions of high loss, such as near the blade tip, the inherent error is magnified. Therefore, the surveys obtained did not give results of sufficient accuracy to provide significant results in comparing the small differences in flow conditions existing between the shrouded- and unshrouded-rotor-blade configurations.

## RESULTS AND DISCUSSION

The results of the investigation are presented in figures 5 to 7.

Over-all performance maps for both the unshrouded- and shrouded-rotor configurations are compared in figure 5. On these maps equivalent turbine shaft work is plotted against the equivalent weight-flow parameter; the ratio of blade speed to design blade speed and the stagnation pressure ratio are shown as parameters together with contours of brake internal efficiency. The design point, corresponding to an equivalent work output of 16.14 Btu per pound and an equivalent mean blade speed of 625 feet per second, is indicated by a circle. The shape of the efficiency contours is quite similar for both configurations, although the efficiency for the unshrouded-rotor configuration is slightly higher over the whole range of the investigation. A direct comparison of the performance of the two configurations can probably be obtained better from figures 6 and 7. In figure 6 the efficiency at design blade speed is plotted against the stagnation pressure ratio. On the average, the efficiency is about one point higher for the unshrouded-rotor configuration over the whole range of pressure ratio. In figure 7 the maximum efficiency obtained at each pressure ratio is plotted in the same manner. Again, the unshrouded-rotor configuration was slightly higher in efficiency over the whole range investigated.

The measured weight flows for the two configurations were the same within 0.50 percent.

For the particular shroud and blade design investigated, the shroud apparently had a slightly adverse effect on the turbine performance. A consideration of the secondary-flow phenomena at the blade tips is necessary to explain the possible cause for the decreased efficiency. In a visualization study of the secondary flow in the tip regions of compressor and turbine blading (ref. 1), it was found that three separate secondary-flow phenomena occur near the blade tips, as discussed in the INTRODUCTION (fig. 1). The tip of the rotor blade was oriented toward the tangential direction from the leading to the trailing edge with a negative entrance angle  $\beta$  so that the adverse effect of the blade-tip scraping would not be appreciable. The forces giving rise to the passage vortex and scraping effect are opposite in direction to the gradient causing the tip-clearance vortex. These pressure gradients may tend to balance out and thereby decrease the over-all secondary flows in the tip region. Such a favorable balance was apparently approached to the degree that the inherent advantages of shrouding were offset by the more favorable secondary flows existing at the tip of the unshrouded-rotor configuration.

## SUMMARY OF RESULTS

A conservatively designed experimental cold-air turbine was investigated both with and without a shroud band on the rotor blades. The tip of the rotor blade was oriented toward the tangential direction so that the adverse effect of the blade-tip scraping would not be appreciable. It was found that the addition of a shroud to the turbine-rotor blades had only a slight effect and did not improve the turbine performance. The efficiency of the unshrouded-rotor configuration was about one point higher than for the shrouded rotor.

Lewis Flight Propulsion Laboratory  
National Advisory Committee for Aeronautics  
Cleveland, Ohio, March 12, 1954

## APPENDIX - SYMBOLS

The following symbols are used in this report:

d	diameter, ft
E	turbine shaft work, Btu/lb
h	specific enthalpy, Btu/lb
p	absolute pressure, lb/sq ft
r	radius, ft
T	total temperature, $^{\circ}$ R
U	blade velocity, ft/sec
w	weight-flow rate of gas, lb/sec
$\eta_t$	brake internal efficiency
$\beta$	relative rotor-blade entrance angle (fig. 1)
$\theta$	temperature reduction ratio, $T/T_0$
$\delta$	pressure reduction ratio, $p/p_0$

Subscripts:

des	design
h	hub
m	mean radius
is	isentropic
T	tip
0	NACA standard sea-level conditions
1	measuring station in surge tank (inlet stagnation condition)
3	measuring station downstream of rotor

Superscript:

'	stagnation state
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## REFERENCES

1. Hansen, Arthur G., Herzig, Howard Z., and Costello, George R.: A Visualization Study of Secondary Flows in Cascades. NACA TN 2947, 1953.
2. Heller, Jack A., Whitney, Rose L., and Cavicchi, Richard H.: Experimental Investigation of a Conservatively Designed Turbine at Four Rotor-Blade Solidities. NACA RM E52C17, 1952.

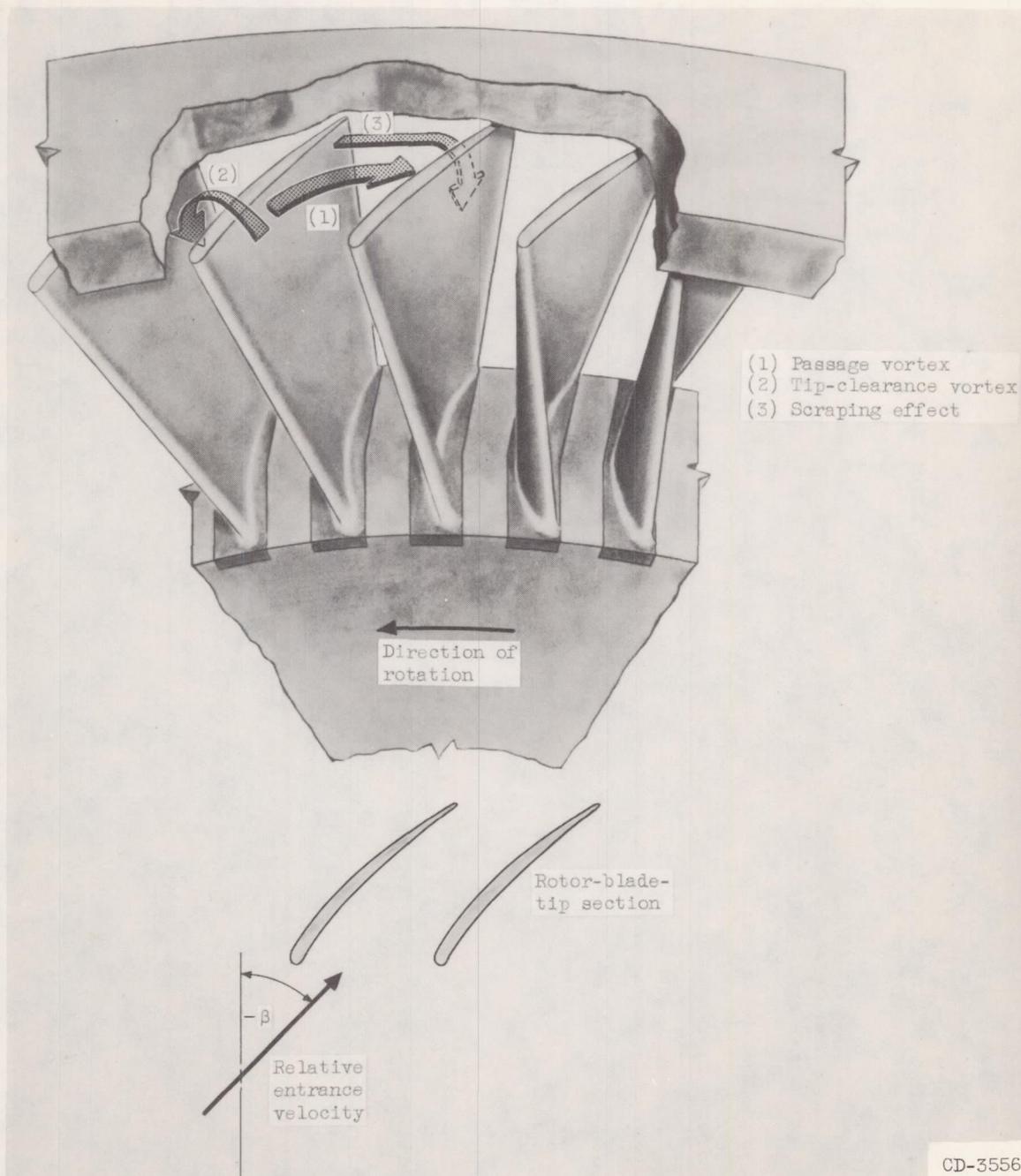
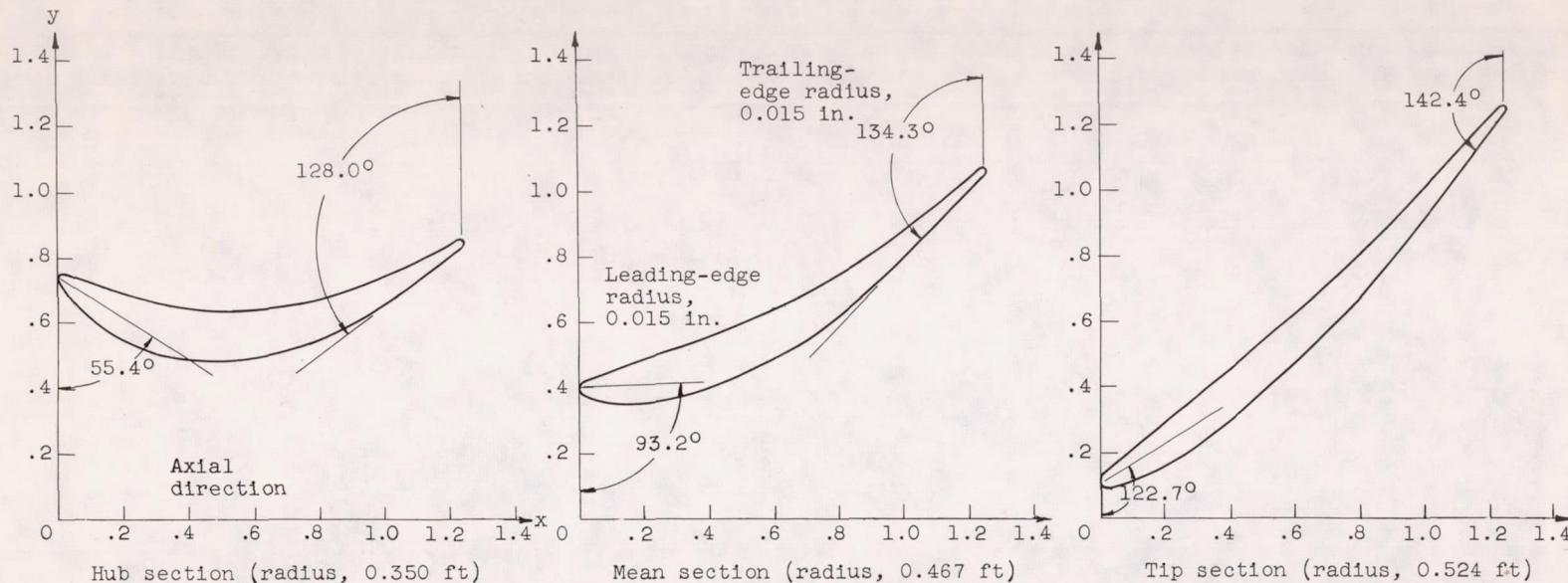


Figure 1. - Diagram of secondary flows at top of turbine-rotor blades.

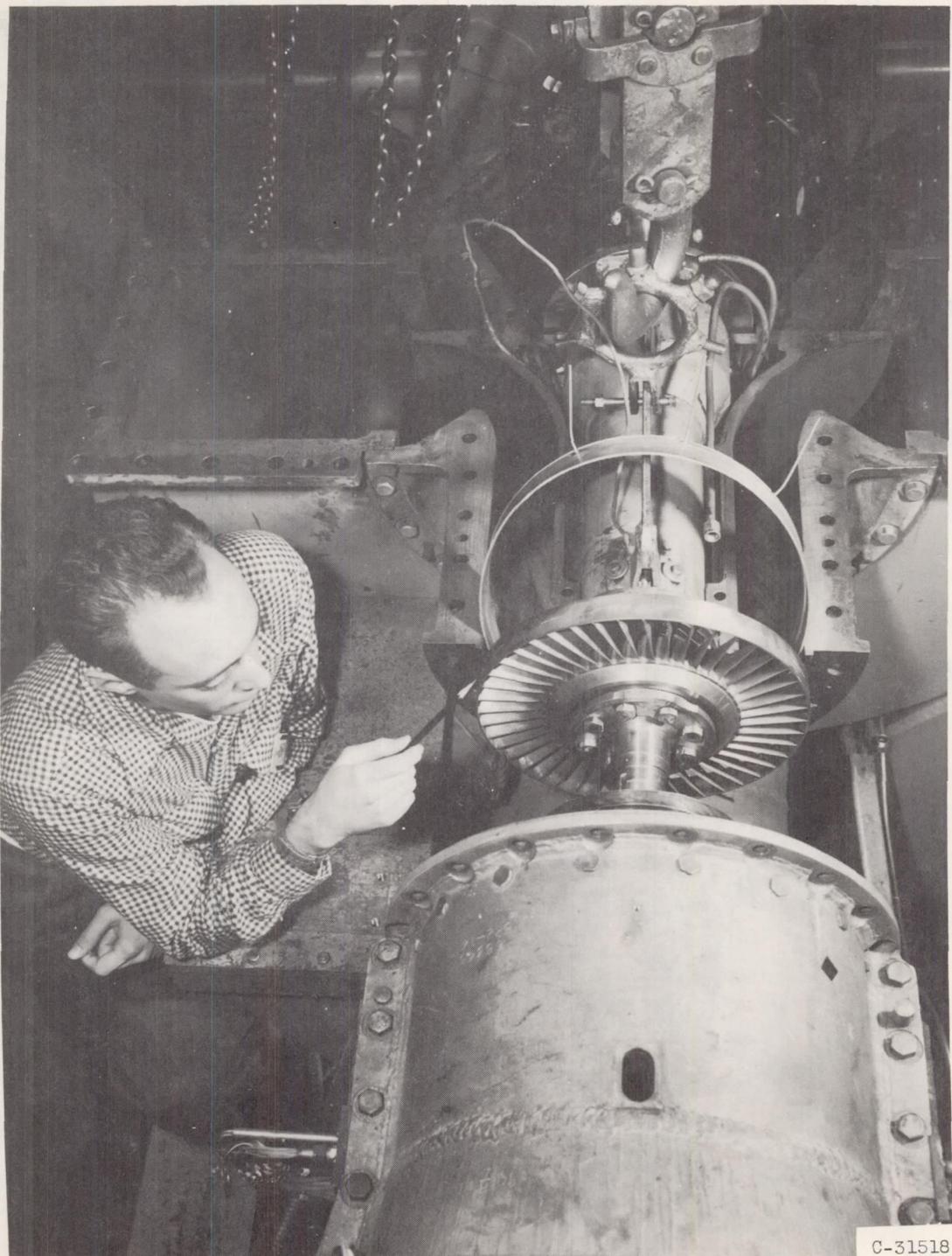


x, in.	y	
	Suction, in.	Pressure, in.
0	0.735	0.735
.1	.616	.725
.2	.549	.687
.3	.512	.661
.5	.489	.637
.7	.516	.654
.9	.594	.701
1.10	.728	.789
1.25	.851	.851

x, in.	y	
	Suction, in.	Pressure, in.
0	0.397	0.397
.1	.362	.451
.2	.359	.486
.3	.372	.521
.5	.448	.598
.7	.546	.690
.9	.702	.808
1.1	.902	.961
1.25	1.063	1.063

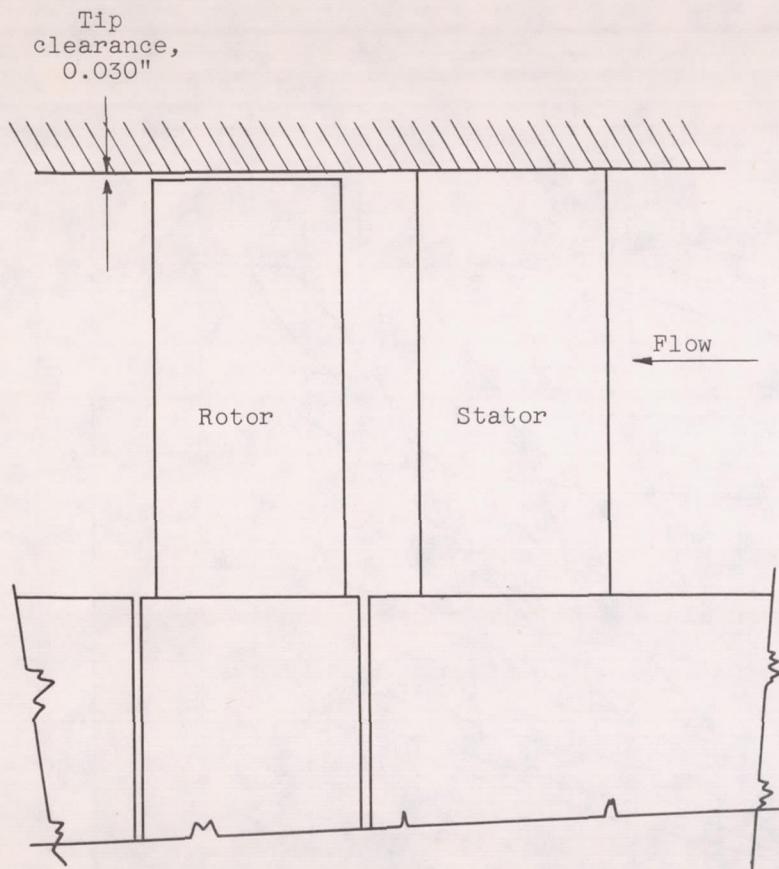
x, in.	y	
	Suction, in.	Pressure, in.
0	0.105	0.105
.1	.114	.215
.2	.164	.298
.3	.230	.377
.5	.388	.541
.7	.574	.713
.9	.801	.900
1.1	1.056	1.110
1.25	1.253	1.253

Figure 2. - Blade-section profiles and coordinates for turbine rotor.

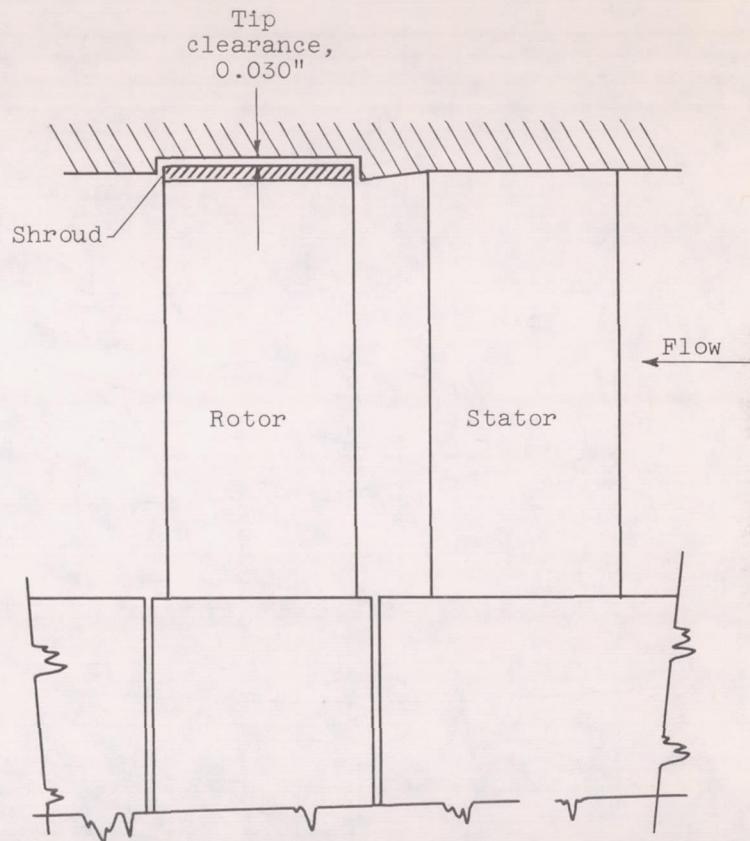


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Figure 3. - Shrouded rotor being installed in turbine test facility.

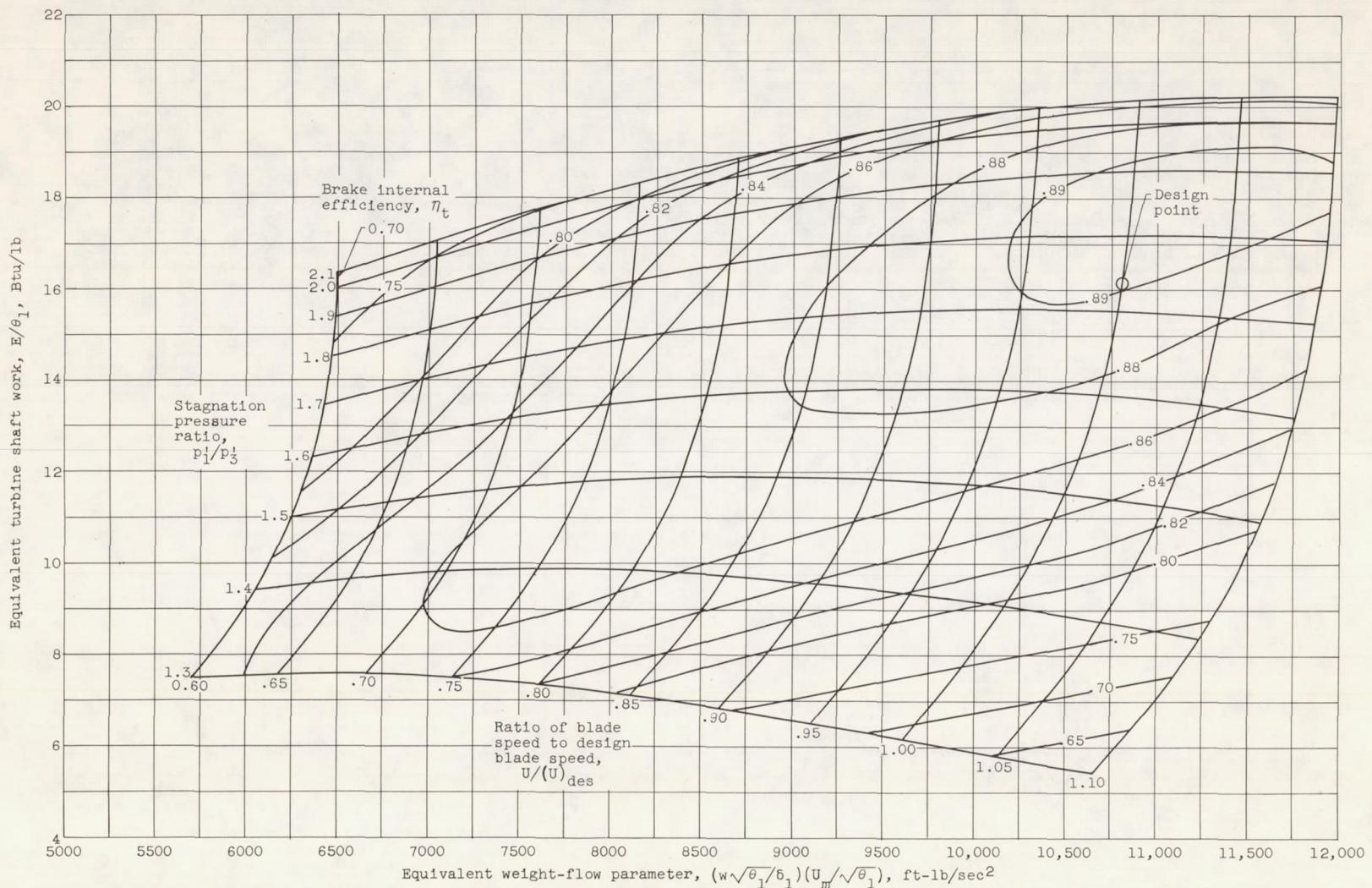


(a) Unshrouded rotor.



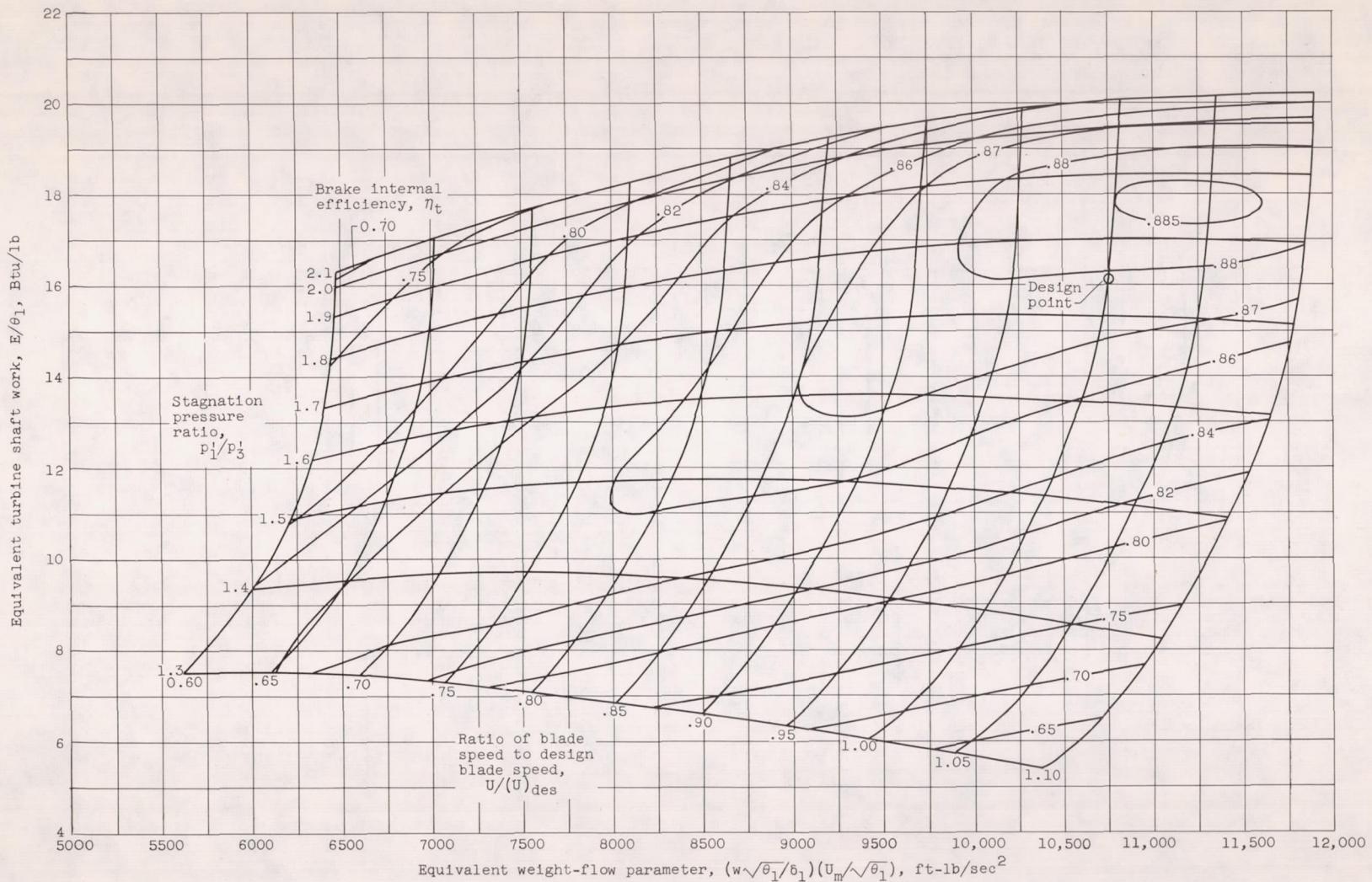
(b) Shrouded rotor.

Figure 4. - Diagrams of shrouded- and unshrouded-rotor configurations.



(a) Unshrouded-rotor configuration.

Figure 5. - Over-all turbine performance.



(b) Shrouded-rotor configuration.

Figure 5. - Concluded. Over-all turbine performance.

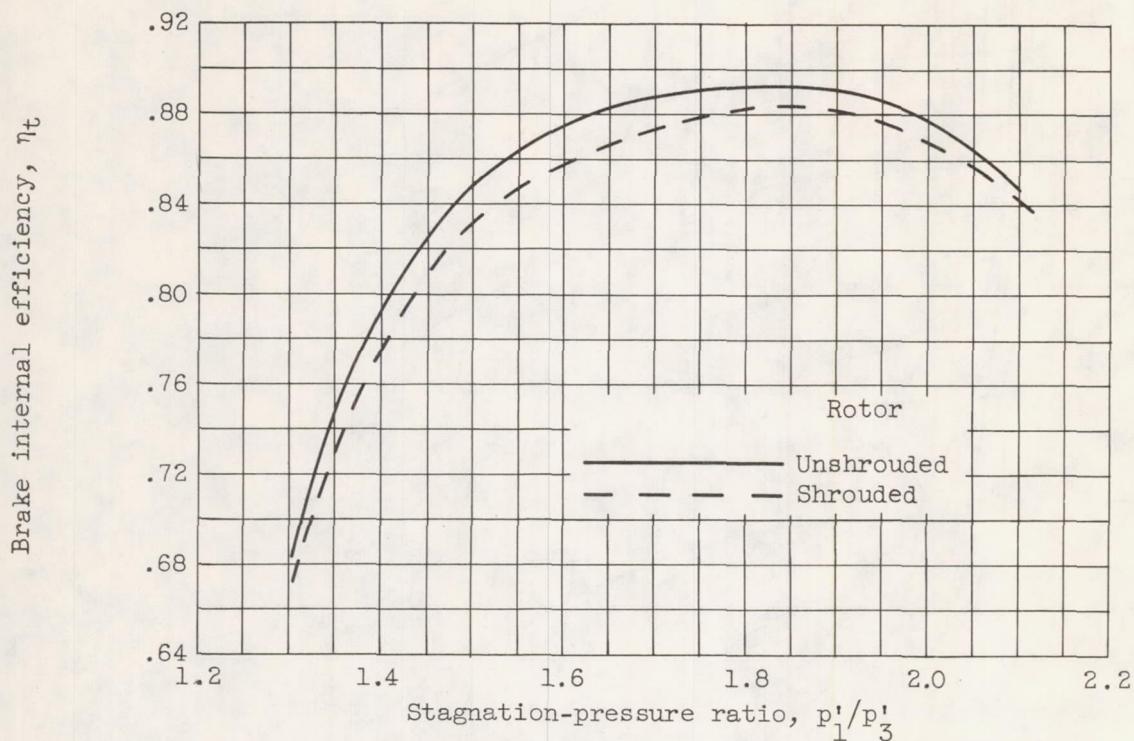


Figure 6. - Effect of rotor shroud on turbine efficiency.  
Ratio of blade speed to design blade speed, 1.0.

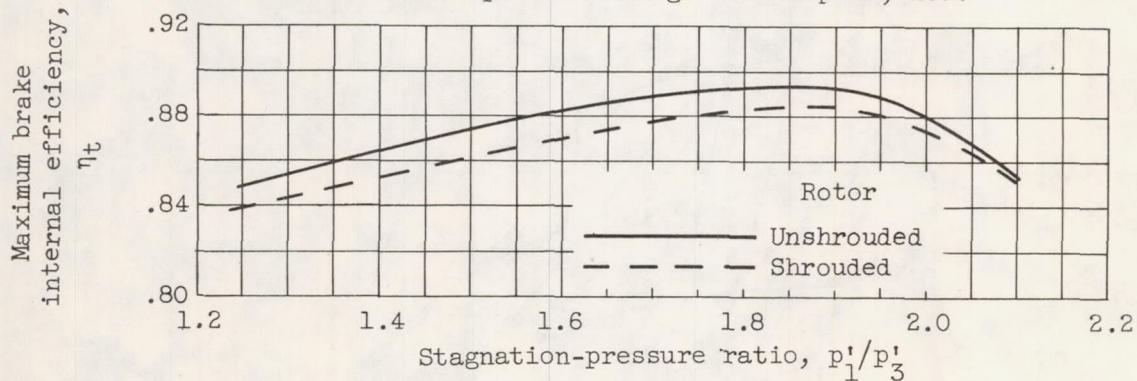


Figure 7. - Effect of rotor shroud on maximum turbine efficiency.